Tire-Wheel-Cavity dynamic model for Structure-borne road noise simulation

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Current tire-wheel models used for structure borne noise simulation are valid up to 200Hz. This limit is mainly due to the lack of tire cavity modes as well as wheel flexibility. Both have a strong effect on forced dynamic responses at wheel center in medium frequency range. Even if the current model is efficient regarding comfort simulation, a new one have to be developed to improve road noise forecast. In the scope of a partnership between PSA Peugeot Citroën and M.F.P.Michelin, the paper deals with the development of a dynamic sub-structured model of a tire wheel assembly -including the air cavity- and its usage for car and tire manufacturers. This new model is a step forward in tire wheel assembly simulation in mid-frequency range, taking into account the interactions between all fluid and structural components. Of course, some new technical challenges already appear that will be discussed: integration of rotating effects, understanding of the ground excitation in the contact patch area.

1 Introduction

In the automotive industry, interior road noise is one of the main issues observed when the tire is rolling on a rough surface. The excitation due to the tire/road contact is transmitted to the cabin through the mechanical properties of the tire and the vehicle.

Interior road noise performance can be splitted into two contributions according to the main physical mechanisms generating the noise, which are:

- up to mid-frequency range, the noise mainly due to solidian transmission through suspensions and car body, which is called structure borne noise
- in high frequency, the interior noise, generated by aerial transmission, named airborne noise.

This paper focuses on interior road noise below 400Hz and only structure borne noise contribution will be considered.

In order to reduce the time and cost of development, the car makers use more and more simulation tools to design their vehicle. Thanks to the improvements done in simulation in the previous years, the vehicle finite element models are more and more realistic. On the one hand, car designers want to use vehicle simulation in dynamic analysis up to 400Hz. On the other hand, the current tire model employed is a dynamic sub structured model valid up to 200Hz. Thus, car makers want a better tire numerical model to simulate efficiently structure borne noise performance taking into account the first air cavity modes and the wheel modes. They need a light model in order to limit memory size of the whole model and have a reasonable computation time. Moreover, tire manufacturers want to protect their industrial and confidential know-how. Finally, a sub structured model seems to be an interesting compromise for all the partners.

This paper deals with the development of a new tire model including the air cavity and flexible wheel modes (cf. [5]) to be representative up to 400 Hz. This model is the result of a collaborative exchange between M.F.P.Michelin and PSA Peugeot Citroën. The first input is a wheel model provided by the car maker. The second input is a tire model and the fluid cavity model under the responsibility of the tire manufacturer. The study consists to generate one sub structured model for each component and to assemble them into a global one. In order to realize these tasks, we have chosen to apply Craig Bampton's sub structuring techniques between structural components (cf. [1]).

First, each model and its associated assumptions is presented. Then, the global sub structured model is described. Finally, model vibro acoustic responses are compared to equivalent measurements.

2 Wheel models

This component is under the responsibility of the car maker. The input is a 3D finite element model of wheel including the rim and the disc. The meshing is created using shell elements for steel wheels and solid elements in the case of aluminium wheels (Figure 1.a). Particular attention should be paid to the junction between rim and disk especially for steel wheels.

Creating a sub structured model of the wheel requires defining precisely its interfaces with other components. The connection with the suspension is done through the bearing face of the wheel on the hub. For frequencies below 400Hz, this zone can be considered as rigid and one node with 6 degrees of freedom is sufficient to describe this interface. To be more consistent with tire symmetry, this node is placed at wheel center defined as the intersection of the wheel median plan and the natural rotation axis of the wheel. All the nodes on the surface of the disc directly connected to the hub are linked to the wheel center by displacement constraints. The wheel center, master node of rigid element, is blocked in all degrees of freedom (Figure 1.b).

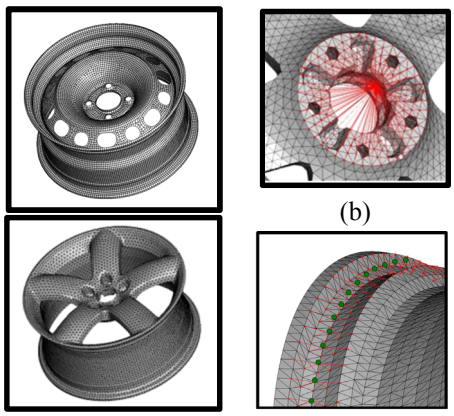


Figure 1: Example of wheel meshing (a), wheel center area (b) and tire/rim interface (c)

Another connection is the natural boundary between tire and wheel. On the [0-400Hz] frequency band studied, some global modal shapes involving rim deformation appear and the interface definition must be able to represent them correctly.

Yet, for each meridian plan, the nodes in contact with the tire can't move easily. The contact pressure between wheel and tire can reach 30 bars locally for a typical automotive tire wheel assembly. Consequently, we assume

that this area moves as a rigid block too. We define a connecting node for each meridian plan and each side of the wheel called tire/rim condensed node. In our case, it is the geometrical barycenter of all wheel nodes of the tire/rim interface in a meridian plan per side. In order to well describe the modal shape of the rim, a refined meshing is necessary. Finally, we have decided to create 180 tire/rim condensed nodes to define the tire/rim interface area, i.e. 90 nodes per side which generate an axisymmetric meshing every 4 degrees (Figure 1.c). That axisymmetric description of rim will be interesting to generate easily the fluid cavity meshing in the next step. In our case, the fluid meshing and structural meshing will be compatible.

The third step is to build the Craig Bampton's sub structured wheel model. The mechanical equations to solve in dynamic analysis are the following (cf. [1], [2] and [4]):

$$(K + j\omega C - \omega^2 M)u = F \qquad (1)$$

where $u = \begin{Bmatrix} u_{int} \\ u_{ext} \end{Bmatrix}$, u_{int} being the internal nodes and u_{ext} the external ones, that is the wheel center and the tire/rim interfaces.

In order to extract the mechanical system, as mentioned in [4], degrees of freedom of external nodes (U_{ext}) and the modal coordinates (α_I) are defined.

The transformation matrix from the initial displacement vector $u = \begin{Bmatrix} u_{int} \\ u_{ext} \end{Bmatrix}$ to the reduced displacement vector $\bar{U} = \begin{Bmatrix} \alpha_I \\ U_{ext} \end{Bmatrix}$ is given as follow:

$$u = \begin{Bmatrix} u_{int} \\ u_{ext} \end{Bmatrix} = \begin{bmatrix} \Phi_{int-int} & A_{INT\_EXT} \\ O & Id_{ext-ext} \end{bmatrix} \begin{Bmatrix} \alpha_I \\ U_{ext} \end{Bmatrix} \qquad (2)$$

where

- $\Phi_{int-int}$ are the eigenvectors of the wheel in blocked conditions
- A_{INT_EXT} are the static modes of the external nodes (wheel center, tire/rim interface)
- $Id_{ext-ext}$ is the identity matrix

Using this transformation, equation (1) becomes:

$$(K_{Wheel}^{DSS} + j\omega C_{Wheel}^{DSS} - \omega^2 M_{Wheel}^{DSS})\bar{U} = \overline{K_{Wheel}^{DSS}}\,\bar{U} = \bar{F} \qquad (3)$$

where

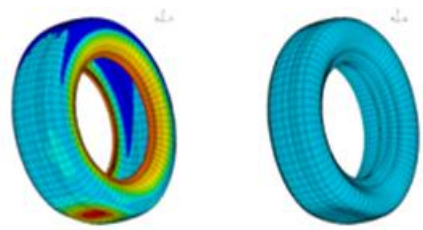
- $\bar{U} = \begin{Bmatrix} \alpha_I \\ U_{Tire/Rim} \\ U_{WC} \end{Bmatrix}$
- $\overline{K_{Wheel}^{DSS}} = \begin{bmatrix} \overline{K_{INT-INT}} & \overline{K_{Int-T/R}} & \overline{K_{Int-WC}} \\ \overline{K_{T/R-INT}} & \overline{K_{T/R-T/R}} & \overline{K_{T/R-WC}} \\ \overline{K_{WC-INT}} & \overline{K_{WC-T/R}} & \overline{K_{WC-WC}} \end{bmatrix}$

3 Tire and cavity models

Under the responsibility of the tire manufacurer, a 3D finite element model of the tire is created taking into account the material and geometrical properties of rubber and tissues: width, thickness, orthotropy, etc. (cf. Figure 2).

In order to simplify the assembly of all components, we define the same condensed nodes on the tire than those on the rim. With an axisymmetrical meshing of the tire, a meridian plan is created every 4 degrees. On the tire/rim interface area, all the tire nodes in contact with the rim are linked to a condensed node by a rigid element.

We can also reduce the number of external nodes in the contact patch. Indeed, this area gives the input of the ground excitation in the tire wheel assembly. On a rough surface, each roughness generates a specific excitation. Several descriptions, more or less complex, can be applied to the contact patch. The most complicated decription defines each tire node included in the contact patch as an external node. The most conventional one reduces the contact patch to one external node called contact patch node, that is the geometrical barycenter of all the nodes included in the contact patch. According to the chosen modelization, the ground excitation applied to the external nodes changes from local excitations to a global one.



(a) (b)

Figure 2: Example of 3D deformed tire meshing (a) and 3D fluid cavity model (b)

First, a non linear computation of the tire is performed as it is mounted on the rim profile which is highly stiff compared to the tire, inflated at its operational pressure and loaded with a static vertical force applied at the wheel center. Obviously, the tire is in a non linear state : large deformation, large displacement, material non-linearities (Figure 2.a).

With the assumptions on external nodes, a Craig Bampton dynamic sub structural tire model is generated between all the external nodes as mentionned in bibliography (cf. [1], [2] and [4]). The relationship between displacement vectors is given by:

$$v = \begin{Bmatrix} v_{int} \\ v_{ext} \end{Bmatrix} = \begin{bmatrix} \Psi_{int-int} & B_{INT\_EXT} \\ O & Id_{ext-ext} \end{bmatrix} \begin{Bmatrix} \beta_I \\ V_{ext} \end{Bmatrix} \qquad (4)$$

where

- v_{int} are internal nodes
- v_{ext} are external nodes (contact patch and tire/rim interface)
- β_I are the modal coordinates
- $\Psi_{int-int}$ are the tire natural modes in blocked conditions
- B_{INT_EXT} are the static of the external nodes (contact patch, tire/rim interface)

The new equation system is the following (5) :

$$(K^{DSS} + j\omega C^{DSS} - \omega^2 M^{DSS})\bar{V} = \overline{K_{Tire}^{DSS}}\bar{V} = \bar{F} \qquad (5)$$

where

- $\bar{V} = \begin{Bmatrix} \beta_I \\ V_{Tire/Rim} \\ V_{CP} \end{Bmatrix}$
- $\overline{K_{Tire}^{DSS}} = \begin{bmatrix} \overline{K_{INT-INT}} & \overline{K_{Int-T/R}} & \overline{K_{Int-CP}} \\ \overline{K_{T/R-INT}} & \overline{K_{T/R-T/R}} & \overline{K_{T/R-CP}} \\ \overline{K_{CP-INT}} & \overline{K_{CP-T/R}} & \overline{K_{CP-CP}} \end{bmatrix}$

From the geometries of inflated, loaded tire and wheel, a 3D surface can be defined on the inner fluid included between the rim and the tire. Introducing fluid properties, the dynamic pressure field p is the result of the following equations in frequency domain :

Material law for adiabiatic fluid:

$$p = -\beta . div(u^f) \qquad (6)$$

Fluid equation of motion:

$$\nabla p = \rho \omega^2 u^f \qquad (7)$$

Where

- u^f is the diplacement field in the fluid
- ρ and β are respectively the fluid density and bulk modulus.

Continuity of the normal displacement at the structural/fluid interface is expressed by:

$$u^f . n = u . n \qquad (6)$$

With a coarse meshing of the fluid cavity (Figure 2.b) defined by the 3D surface, a modal analysis of the fluid can be performed to compute the eigenvalues (Ω_f^2) and the natural modes (Γ_f) of the fluid. According to [3] and [5], the fluid/structural coupling terms at the borders between the cavity and, on the one hand the tire (Γ_f^{Tire}), on the other hand, the rim (Γ_f^{Wheel}) can be computed taking into account the stiffness of the fluid on the surface and the pressure field in the fluid expressed by the eigen vectors (Γ_f).

The transformation matrix from the initial pressure field p to the reduced vector $\bar{P} = \{\gamma_k\}$ is given by the following formulae:

$$p = [\Gamma_f]\{\gamma_k\} \qquad (7)$$

4 Assembly of the Tire Wheel Cavity (TWC) Model

After computing each component, the assembly of the models can be performed taking into account the reduced displacement vector X which combines the wheel reduced vector, the tire reduced vector and the fluid vector:

$$\bar{X} = \begin{Bmatrix} \gamma_k \\ \alpha_I \\ \beta_J \\ U_{Tire/rim} \\ U_{WC} \\ V_{CP} \end{Bmatrix}$$

Thus, according to patent [5], the global equation system (10) to solve is the following:

$$\overline{K_{TWC}^{DSS}} \bar{X} = \bar{F} \qquad (8)$$

Where

$$\overline{K_{TWC}^{DSS}} = \begin{bmatrix} \Omega_f^2 & \Psi_{tire}\Gamma_f^{Tire} & \Phi_{wheel}\Gamma_f^{Wheel} & 0 & B\Gamma_f^{Tire} & A\Gamma_f^{Wheel} \\ (\Psi_{tire}\Gamma_f^{tire})^t & \overline{K_{INT-INT}^{Tire}} & 0 & \overline{K_{INT-T/R}^{Tire}} & \overline{K_{INT-CP}^{Tire}} & 0 \\ (\Phi_{wheel}\Gamma_f^{wheel})^t & 0 & \overline{K_{INT-INT}^{TireWheel}} & \overline{K_{INT-T/R}^{Wheel}} & 0 & \overline{K_{INT-WC}^{Wheel}} \\ 0 & \overline{K_{T/R-INT}^{Tire}} & \overline{K_{T/R-INT}^{Wheel}} & (\overline{K_{T/R-T/R}^{Tire}} + \overline{K_{T/R-T/R}^{Wheel}}) & \overline{K_{T/R-CP}^{Tire}} & \overline{K_{T/R-CP}^{Wheel}} \\ (B\Gamma_f^{tire})^t & \overline{K_{CP-INT}^{Tire}} & 0 & \overline{K_{CP-T/R}^{Tire}} & \overline{K_{CP-CP}^{Tire}} & 0 \\ (A\Gamma_f^{wheel})^t & 0 & \overline{K_{WC-INT}^{Wheel}} & \overline{K_{WC-T/R}^{Wheel}} & 0 & \overline{K_{WC-WC}^{Wheel}} \end{bmatrix}$$

The sub structural tire wheel cavity model is a square matrix (about 1200 x 1200) taking into account:

- The first natural modes of the fluid cavity
- One hundred natural modes of the tire
- One dozen natural modes of the wheel
- About 1000 dofs of connecting nodes at the tire/rim interface
- 6 dofs at the wheel center node
- 6 (or more) dofs at the contact patch node(s)

If necessary, the global size of the matrix can be reduced by a new sub structuring method after sorting the system into two groups: internal nodes where no external excitation can be applied and external nodes where it is possible to apply excitation or link to other mechanical system.

$$u = \begin{Bmatrix} u_{int} \\ u_{ext} \end{Bmatrix} = \begin{bmatrix} \Pi_{int-int} & C_{INT\_EXT} \\ O & Id_{ext-ext} \end{bmatrix} \times \begin{Bmatrix} \delta_I \\ U_{ext} \end{Bmatrix} \qquad (9)$$

The conventional reduction performed for the car industry is the following:

- A group of internal degrees of freedom including the modal coordinates of the cavity, the modal coordinates of the wheel, the modal coordinates of the tire and the connecting nodes at the tire /rim interface
- A group of external degrees of freedom including the wheel center node and the contact patch node(s)

According to this new description, the global size of the sub structured tire wheel cavity model is similar to that of the current tire model including the mechanical matching between components by a square matrix (about 150*150). It is composed by the first N natural modes of the tire wheel cavity system, 6 degrees of freedom at the wheel center and 6 (or more) degrees of freedom in the contact patch. After writing this tire wheel cavity model in a text format like NASTRAN, it can be used directly in a simulation chain as the current tire model without any further specification.

5 Validation and application in the automotive industry

In order to validate the tire wheel cavity model, a first step consists in assessing each component individually. In the case of the wheel, an experimental modal analysis can be performed according to the following boundary conditions:

- Free tire/rim interface area
- Blocked wheel center area as the wheel was mounted on a rigid hub

A specific test bench (cf. Figure 3) was used to obtain the results without any interference in the frequency band of interest.

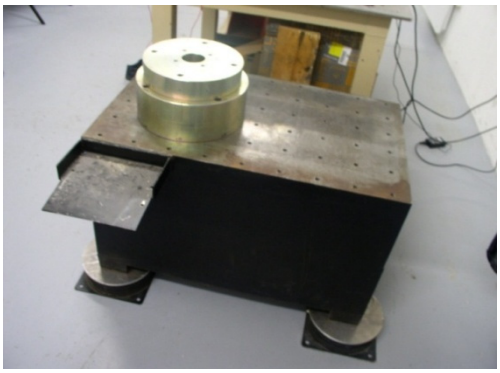


Figure 3: Experimental wheel test bench

The correlation obtained between the sub structured model and the measurements is quite good (cf. Table 1).

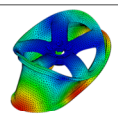
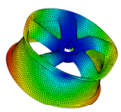
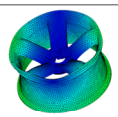
Mode					
	ovalling	ovalling 45°	bending	bending 90°	pumping
Test	418	421	436	443	642
Simulation	430	432	442	443	658
Error (%)	-2,9%	-2,7%	-1,3%	-0,1%	-2,5%

Table 1: Example of measured/computed wheel modes

In the tire wheel assembly, the stiffness of the wheel is higher than these of the rubber and the fluid. Thus, the mechanical behavior of this component in the assembly is close to one measured individually. The same assumptions can't be applied neither for the tire nor the fluid. Thus, the second step to validate the dynamic sub structured model consists in frequency forced responses on the tire wheel assembly.

In order to be close to the usage, we have decided to evaluate the dynamic stiffness of the tire wheel assembly between the contact patch and the wheel center. Indeed, there are transfer functions between the ground excitation area and the connecting area with the car body. Moreover, the external nodes in the sub structural model are the same. In order to validate the sub structural model using transfer functions, it is essential to know the excitation function in the contact patch and to measure correctly the forces and torques at the wheel center.

The test bench used to identify tire natural modes is depicted on Figure 4. The ground is represented by a rigid flat sheet on bearings. On the one hand, a displacement in a canonical direction is applied to the sheet by a dynamic shaker, an accelerometer on the sheet recording the movement. On the other hand, the wheel center is rigidly linked to the test bench. Thanks to the force cell responses on the bench, a global tensor (forces and torques) at the wheel center is evaluated. We can obtain the FRF between the tensor and the accelerations to evaluate the dynamic stiffness matrix of the tire wheel assembly. Nevertheless, the quality of this measurement is directly related to the stiffness of the test rig. The target was to perform measurements up to 400 Hz. But, the first natural modes on this test rig appeared above that frequency. So, the experimental dynamic stiffness could be disturbed by these modes, especially in the higher frequency range.

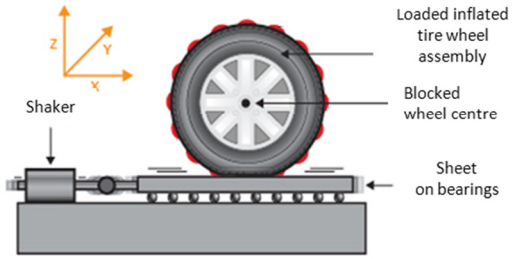


Figure 4: Scheme of experimental test rig in one direction

In the simulation, some canonical displacements can be applied in the frequency domain at the contact patch, the other degrees of freedom being, for example, blocked. If the wheel center is fixed, an evaluation of forces and torques at the wheel center generated by displacement in the contact patch can be evaluated. These computations simulate the experiments on the test bench.

These simulations can be performed with all kinds of tire wheel assembly models. Here, we show the computation obtained with the current tire model and the new tire wheel cavity model compared with the measurement. Even if the measurement is disturbed by the mechanical behavior of the test rig, the tire wheel cavity model is more accurate than the current model.

An example of results is presented in Figure 5. The dynamic stiffness in Z direction is evaluated. Given the tire dimension, the first cavity mode is evaluated about 230 Hz. The sub structured model (TWC) gives a good estimation of the dynamic stiffness in this frequency range.

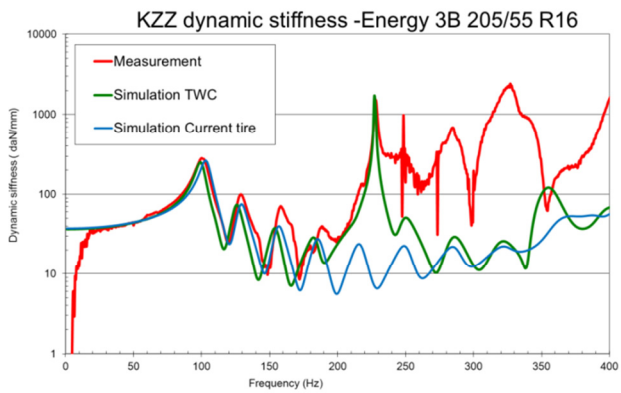


Figure 5: Example of comparison between simulated and measured dynamic stiffness

The new tire wheel assembly model can be used in a vehicle simulation as the current tire model in dynamic analysis. Today, this linear model can be employed in the field of the comfort performance to simulate small obstacles on road, like cleats. Connecting it to the vehicle hub, it is possible to evaluate all kinds of vehicle transfer function, acoustical or vibrational ones, between the contact patch or the wheel center and any other point in the vehicle simulating accelerometers or microphones.

Finally, it is possible to assess every tire wheel assembly characteristics, especially between the contact patch and the wheel center as expected by the tire manufacturer. For car makers, it is essential to simulate properly the vehicle transfer function between the tire wheel assembly (contact patch, rim seats, wheel center or any other point) and the vehicle. Nevertheless, this linear model doesn't take into account neither large displacements nor large strains on the tire wheel assembly.

6 Conclusion

This paper reports the development of a new model for the tire wheel assembly. It is based on sub structuring techniques to generate a black box element taking into account the mechanical behavior of tire, wheel and fluid cavity and their interactions together. In fact, each industrial partner (car maker or tire manufacturer) cares for his own component. The technique to generate a tire wheel cavity model is mentioned in patent [5].

This model is a step forward for the automotive industry to evaluate the tire wheel assembly characteristics and the vehicle transfer functions. Some validations and applications in the automotive industry are illustrated to demonstrate the improvements made in tire models. Nevertheless some future challenges already appear at this stage. First point, how to implement rotating effect in simulation? Indeed, the structure borne performance is evaluated in rolling conditions. And the mechanical behavior of the tire wheel assembly is influenced by rolling effects (split of the cavity modes for example). Second point, how to simulate ground excitation in the contact patch area? When these new challenges will be solved, a consistent simulation chain of structure borne interior road noise will be achieved and time and cost reduction will have reached a new optimum.

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